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DYNAMIC ELASTO-PLASTIC RESPONSE OF SHELLS IN AN ACOUSTIC MEDIUM

THEORETICAL DEVELOPMENT FOR THE EPSA CODE .

by

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ABSTRACT

The "EPSA" (Elasto-Plastic Shell Analysis) code has been developed for the analysis of shells in an acoustic medium subjected to dynamic loadings which produce large elasto-plastic deformations in the shell. The analysis includes the modeling of significant internal structures, which produce hard spots on the shell. In addition, the effects of ambient pressure are considered. This report presents the theoretical development for the "EPSA" code and a description of the code itself. A users manual for "EPSA" is planned for the future.

The structural equations of motion are derived from the principle of virtual work and descretized over the shell in a manner typical of finite element procedures. The integration in time of the equations of motion are done explicitly via a central difference scheme.

The nonlinear Donnell-Vlasov kinematic equations of shell theory are used. Plate strain-displacement relations are established by a two dimensional finite difference scheme.

Two special features have been incorporated into "EPSA" in order to obtain a major gain in the efficiency of the calculations. First, a self consistent plasticity theory for shells has been developed directly in terms of the stress resultants thereby avoiding conventional "through-the-thickness" integrations. Second, a modification of the basic quadrilateral element has been made using finite difference techniques in which the rotational degrees of freedom are removed from the nodal points. As described in the report, both procedures result in a marked increase in computational efficiency, particularly for cases in which large systems are to be analyzed.

The fluid-structure interaction is accounted for by means of the Doubly Asymptotic Approximation (DAA) expressed in terms of orthogonal fluid expansion functions.

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NOMENCLATURE

$\{a\}$	Vector of derivatives of w
A	Parameter in flow rule equation
$A_i^{(k)}$	k^{th} sub-area of element i
[A]	Matrix of nodal point coordinates
[AA]	Matrix relating nodal point displacements to derivatives of w
$[B_k]_i$	Strain-displacement matrix for the k^{th} region of the i^{th} element
c	Sound velocity in fluid
[D]	Tangent moduli matrix
$\{e_k\}_i$	Strain vector in the k^{th} region within element i
e_{xx},e_{yy},e_{xy}	Components of strain in Cartesian coordinates; components of vector $\{e_k\}$
e_{11},e_{22},e_{12}	Components of strain in orthogonal coordinates
$\{e_k''\}$	Plastic component of strain vector
[D]	Elasto-plastic tangent stiffness matrix
[E]	Elastic moduli matrix
F, F_o, F_L	Current, initial and limit yield function
F_S, F_M	Absolute values of the gradients of the yield function F
$\{F\}_i$	Force vector in element i
h_1,h_2	Metric coefficients
I_N,I_M,I_{NM},I_M^*	Stress resultant invariants
[J]	Jacobian matrix
k_{xx},k_{yy},k_{xy}	Components of curvature in Cartesian coordinates; components of vector $\{e_k\}$
K_{11},K_{22},K_{12}	Components of curvature in orthogonal coordinates
M	Total number of surface expansion functions
[M]	Lumped mass matrix

M_{xx}, M_{yy}, M_{xy}	Shell moments per unit length; components of the stress resultant vector {s}
M_{ij}^{*}	Hardening parameters for moments (residual moments)
M_o	Initial yield moment for shell
N_o	Initial yield force for shell
N	Total number of elements within the structure
{N}	Interpolation (shape) function
N_{xx}, N_{yy}, N_{xy}	Shell normal force per unit length; components of {s}
{P}	External force vector acting on structural nodes
{p}	Surface loading per unit area
$\{q\}_i$	Nodal displacement vector of element i
{Q}	Generalized fluid force vector
R_x, R_y	Radii of curvature along the principal directions of the shell in Cartesian coordinates
R_1, R_2	Radii of curvature along the principal directions of the shell in orthogonal coordinates
{S}	Stress resultant vector
{S*}	Stress resultant vector including residual moments
s,t	Local coordinate system axes
t_i	i^{th} time step
{U}	Displacement vector for a point within an element
u_1, u_2, u_3, u_4	Tangential displacements in the x direction of the four nodes contiguous to an element; components of vector {q}
{V}	Incident fluid velocity vector
V_I	Incident fluid velocity at structural node i
v_1, v_2, v_3, v_4	Tangential displacements in the y direction of the four nodes contiguous to an element; components of vector {q}
$w_1, w_2, \ldots . w_{12}$	Normal displacements of the 12 nodes surrounding an element; components of vector {q}

x, y	Cartesian coordinates
$\{Y\}$	Integral in time of generalized fluid forces
$[\gamma]$	Inverted virtual mass matrix
$[\Psi]$	Surface expansion function matrix
Δt	Time step
α	A parameter in the yield condition
λ	A parameter in the flow rule
$\varepsilon_1 : \varepsilon_2$	Orthogonal coordinates along the principal curvatures
ρ	Mass density per unit area of shell surface
ρ_f	Mass density of fluid
ϕ_1, ϕ_2	Rotations with respect to ε_1 and ε_2
σ_{ij}	Stress components
σ_o	Yield stress in uniaxial tension
ν	Poisson ratio
μ_k	Wet surface area of shell

I INTRODUCTION

In recent years, considerable progress has been made in the development of methodology and computer codes for the analysis of the response of structures in an acoustic medium under dynamic loadings. Generally, two types of problems have been studied: (1) the analysis of structures in which both the basic shell structure and the internal components act elastically under the dynamic loadings [1], [2]; and (2) the analysis of structures which undergo large elasto-plastic deformations under the dynamic loadings.

This report is concerned with the analysis of large elasto-plastic deformations of stiffened shells under dynamic loadings. Previous reports by Weidlinger Associates that have been issued on this subject include Ref. [3] which is concerned with the development of an elasto-plastic theory for the analysis of stiffened shells and Ref. [4] which provides an overview of the numerical procedures that have been employed on the dynamic analysis of elasto-plastic shells.

The complexity of the problems encountered in developing methodology for the elasto-plastic analysis of such shells with internal structure requires that a numerical discretization procedure be utilized. For this purpose, the "EPSA" (Elasto-Plastic Shell Analysis) code has been developed. "EPSA" is a modified finite element code which incorporates a number of specific features which are geared to the efficient analysis of submerged shells subjected to shock loadings.

The specific objectives which guided the development of "EPSA" are as follows:

1) Analysis of shells in an acoustic medium subjected to both high and low

frequency shock loadings.

2) Efficient modeling of elasto-plastic behavior.

3) Inclusion of large displacement effects to analyze dynamic buckling situations and post-buckling behavior.

4) Efficient treatment of the structure-media interaction problem.

5) Modeling techniques which can be used for complex shells of arbitrary geometry, including stiffeners and internal structures. The internal structures often produce "hard spots" which materially affect the motions of the shell structure.

In a code which is to be a useful analysis/design tool, these objectives must be met in an efficient fashion to facilitate its application to the solution of large problems involving complex structures.

The objective of this report is to present the basic theory and methodology upon which the EPSA code is based. The specific features and aspects of "EPSA" will be discussed in detail in the Sections which follow. Two of these features have been utilized to obtain a major gain in the efficiency of the calculations (both with respect to accuracy and to running time), and are worthy of special mention at this time. First, an elasto-plastic shell theory, defined in terms of the moments and direct force resultants on the shell, has been developed, Ref. [3], and utilized in EPSA. This is done in place of the usual "through-the-thickness" integration techniques for elasto-plastic analysis. The use of the elasto-plastic theory results in a material increase in the efficiency of the computational procedure particularly when applied to complicated problems.
The second special feature involves a modification of the basic quadrilateral element by means of finite difference techniques, in which the rotational degrees of freedom are removed from the nodal points. This results in a modified twelve noded, twenty degree of freedom element in which the degrees of freedom are all displacements.

Several advantages are gained by the elimination of the rotational degrees of freedom. First, the set of differential equations to be solved in time is reduced since only the translational degrees of freedom appear at each node. Second, the present quadrilateral element results in much simpler computations than in any generally used conforming finite element. In addition, one avoids the possibility of an ill conditioned mass matrix and the related unduly stringent stability limitations associated with the high frequencies produced by the rotational mass terms.

The inclusion of the aforementioned special features makes "EPSA" especially useful for the analysis of complicated structures, and in particular those with significant hard spots introduced by internal equipment.

Comparisons of "EPSA" results with both analytical and experimental results for a series of problems involving the dynamic loading of elasto-plastic shells in vacuo are presented in this report.

II. EQUATIONS OF MOTION OF THE STIFFENED SHELL

The equations of equilibrium are written in the form of the principle of virtual work:

$$\int_R \{s\}^T\{\delta e\}\, dR - \int_R \{p\}^T\{\delta U\}\, dR + \int_R \rho\{\ddot{U}\}\{\delta U\}\, dR = 0 \tag{2.1}$$

with,

$$\{U\} = (u_1, u_2, w)^T$$

$$\{s\} = (N_{11}, N_{22}, N_{12}, M_{11}, M_{22}, M_{12})^T$$

$$\{e\} = (e_{11}, e_{22}, 2e_{12}, k_{11}, k_{22}, 2k_{12})^T$$

The scheme used to solve the above variational problem is as follows;

The surface of the region is covered by a quadrilateral mesh, each element of which has an area A_i. The integrals over R are replaced by the sums of integrals over A_i. The summation over A_i in turn is performed by subdividing the element into regions $A_i^{(k)}$ as shown in Fig. 1. Therefore,

$$\int_R \{s\}^T\{\delta e\}\, dR = \sum_{i=1}^{N}\sum_{k=1}^{4} \{s\}_i^T\{\delta e_k\}_i A_i^{(k)} =$$

$$\sum_{i=1}^{N}\sum_{k=1}^{4} \{s\}_i^T[B_k]_i\{\delta q\}_i A_i^{(k)} = \{F\}^T\{\delta q\} \tag{2.2}$$

$$\int_R \{p\}^T\{\delta U\}\, dR = \sum_{i=1}^{N}\sum_{k=1}^{4} \{p\}^T\{\delta q\}_i A_i^{(k)} = \{P\}^T\{\delta q\} \tag{2.3}$$

$$\int_R \rho\{\ddot{U}\}^T\{\delta U\}\, dR = \sum_{i=1}^{N}\sum_{k=1}^{4} \rho\{\ddot{q}\}_i^T\{\delta q\}_i A_i^{(k)} = [M]\{\ddot{q}\}^T\{\delta q\} \tag{2.4}$$

where $\{q\}$ is the nodal displacement vector for the structure, [M] is the lumped mass matrix, {P} represents the vector of external forces acting on the nodes of the structure, {F} is the vector of equivalent internal grid point forces and N is the number of elements.

The principle of virtual work is therefore transformed to the following system of ordinary differential equations:

$$[M]\ \{\ddot{q}\} = \sum_{i=1}^{N} [\{P\}_i - \{F\}_i] \tag{2.5}$$

The general solution procedure for Eq. (2.5) is described in Section III. An explicit numerical scheme is utilized for the forward step integration in time.

III GENERAL SOLUTION PROCEDURE

The system of equations for the nodal displacements of the structure, Eq. (2.5) is integrated in time utilizing the following central difference scheme which is an explicit method. For the solution of problems involving the treatment of shock waves across the structure, accuracy requirements preclude the use of large time steps. For such problems explicit time integration methods are particularly optimal. The velocity of node i at time step t_{j+1} is computed by

$$\{q\}_i^{t_{j+1}} = \{q\}_i^{t_j} + \frac{\Delta t}{M_i}\left(\{P\}_i - \sum_{k=1}^{4} \{F\}_k\right) \tag{3.1}$$

where element forces $\{F\}_k$ are summed over all elements k framing into node i, M_i is the mass of the node, and $\{P\}_i$ are the externally applied forces on the nodes.

The following procedure is employed at each time step of the solution phase for each element of the shell,

i) Nodal point velocities are updated by the application of external loads occurring within the time step. These loads are time varying in either functional or discrete form.

ii) Boundary conditions are enforced where applicable.

iii) The strain increment occurring within the time step is computed from the nodal point displacements via the strain-displacement relations.

iv) The increment in the element stress state is calculated by use of the constitutive equations.

v) Internal stress state is converted into equivalent grid point forces.

vi) Nodal point velocities are updated by the application of grid point forces occurring within the time step.

Note that the formulation of the equations is in the initial configuration. All equations are solved in the initial geometry and all references are to the initial coordinate directions and initial areas (Lagrangian).

IV KINEMATIC EQUATIONS

The Donnell-Vlasov nonlinear kinematic equations of shell theory are employed. In orthogonal coordinates, the strain-displacement relations read

$$
\begin{aligned}
e_{11} &= \frac{\partial u_1}{h_1 \partial \xi_1} + \frac{1}{h_1 h_2} \frac{\partial h_1}{\partial \xi_2} u_2 + \frac{w}{R_1} + \underline{\frac{1}{2} \phi_1^2} \\
e_{22} &= \frac{\partial u_2}{h_2 \partial \xi_2} + \frac{1}{h_1 h_2} \frac{\partial h_2}{\partial \xi_1} u_1 + \frac{w}{R_2} + \underline{\frac{1}{2} \phi_2^2} \\
2e_{12} &= \frac{\partial u_2}{h_1 \partial \xi_1} + \frac{\partial u_1}{h_2 d \xi_2} - \frac{1}{h_1 h_2} \frac{\partial h_1}{\partial \xi_2} u_1 - \frac{1}{h_1 h_2} \frac{\partial h_2}{\partial \xi_1} u_2 + \underline{\phi_1 \phi_2} \\
k_{11} &= \frac{\partial \phi_1}{h_1 \partial \xi_1} + \frac{1}{h_1 h_2} \frac{\partial h_2}{\partial \xi_1} \phi_2 \qquad k_{22} = \frac{\partial \phi_2}{h_2 \partial \xi_2} + \frac{1}{h_1 h_2} \frac{\partial h_2}{\partial \xi_2} \phi_1 \\
2k_{12} &= \frac{\partial \phi_2}{h_1 \partial \xi_1} + \frac{\partial \phi_1}{h_2 \partial \xi_2} - \frac{1}{h_1 h_2} \frac{\partial h_1}{\partial \xi_2} \phi_1 - \frac{1}{h_1 h_2} \frac{\partial h_2}{\partial \xi_1} \phi_2
\end{aligned}
\tag{4.1}
$$

where

$$
\phi_1 = - \frac{\partial w}{h_1 \partial \xi_1} \qquad \phi_2 = - \frac{\partial w}{h_2 \partial \xi_2} \tag{4.2}
$$

The underlined terms represent geometric nonlinearities.

V DISCRETIZATION

An arbitrarily shaped structure is divided into its constitutive parts called "sheets". Each sheet is a curved section of shell with an arbitrary number of nodes and elements (Fig. 2). The shape of the sheet is limited to a surface that can be described by a smooth continuous function without any interior discontinuities in its slope. There can be no corners or edges within a sheet.

Thus a cylinder with flat end caps would consist of three sheets a circular cylindrical sheet and a planar sheet for each end cap (Fig. 3). Three sheets are required to specify the structure because of the edge that occurs between the cylinder and the flat end caps.

Multi-sheet capability consists of assuring the following boundary conditions on the nodes along a shared edge:

1) compatibility of displacements of nodes along the edge.

2) compatibility of rotations of nodes along the edge.

3) equilibrium of moments at the nodes along the edge.

Each sheet is further sub-divided into elements. The elements within a sheet can be of any arbitrary quadrilateral organization.

Discrete stiffener elements are available (Ref. [4]) to model any stiffeners which exist on the shell.

Each arbitrarily shaped quadrilateral shell element is defined by four corner nodes, with each node having three translational and no rotational degree of freedom terms*). In order to represent bending behavior (second derivative terms) eight nodes not contiguous with the element are also used (Fig. 4).

Each element accesses twelve nodes and has twenty degrees of freedom: three translational degrees of freedom for each of the four inner nodes, and one degree of freedom (displacement normal to the surface) for each of the eight exterior nodes. Thus, the nodal displacement vector of an element i is

$$\{q\}_i = (u_1, u_2, u_3, u_4, v_1, v_2, v_3, v_4, w_1, w_2, w_3, w_4, w_5 \ldots w_{12}) \qquad (5.1)$$

For improved accuracy, each element is further divided into four regions for the computation of strains in each region.

The strain in element i is

$$\{e\}_i = (\{e_1\}, \{e_2\}, \{e_3\}, \{e_4\})_i$$

where e_1 is the strain in region 1 and

$$\{e_1\} = (e_{xx}, e_{yy}, 2e_{xy}, K_{xx}, K_{yy}, 2K_{xy})^T \qquad (5.2)$$

The element strains are related to the nodal displacements as follows;

$$\{\delta e_k\}_i = [B_k]_i \{\delta q\}_i = \{[B_k]_i' + [B_k]_i''\} \{\delta q\}_i \qquad (5.3)$$

*) "EPSA's" element formulation differs from that of conventional finite element codes where rotations and curvatures are considered as additional degrees of freedom and interpolating functions are used to establish the strain-displacement relations.

[B]' is a function solely of the element geometry. [B]" is a function of both the element geometry and the normal displacements. [B]" represents the nonlinear terms in the strain displacement relations which are the extending terms for the range of moderately large deflections. These nonlinear terms account for the finite displacement gradients $\partial w/\partial \xi_1$ and $\partial w/\partial \xi_2$ in the strain-displacement relations. $\{\delta e\}$ and $\{\delta q\}$ are the strain and displacement increments.

The Donnell-Vlasov nonlinear kinematic equations relating strain and displacement increments, written in Cartesian coordinates are:

$$\begin{aligned}
\delta e_{xx} &= \frac{\partial \delta u}{\partial x} + \frac{\delta w}{R_x} + \underline{\frac{\partial w}{\partial x}\frac{\partial \delta w}{\partial x}} \\
\delta e_{yy} &= \frac{\partial \delta v}{\partial y} + \frac{\delta w}{R_y} + \underline{\frac{\partial w}{\partial y}\frac{\partial \delta w}{\partial y}} \\
2\delta e_{xy} &= \frac{\partial \delta v}{\partial x} + \frac{\partial \delta u}{\partial y} + \underline{\frac{\partial w}{\partial y}\frac{\partial \delta w}{\partial x} + \frac{\partial w}{\partial x}\frac{\partial \delta w}{\partial y}} \\
\delta k_{xx} &= \frac{\partial^2 \delta w}{\partial x^2}, \quad \delta k_{yy} = \frac{\partial^2 \delta w}{\partial y^2}, \quad \delta k_{xy} = \frac{\partial^2 \delta w}{\partial x \partial y}
\end{aligned} \tag{5.4}$$

where the underlined terms account for finite displacement effects.

First derivative (membrane) terms of the [B] matrix are computed by mapping the quadrilateral element via the Jacobian into a local s-t coordinate system where a linear shape function is assumed.

$$\begin{bmatrix} \frac{\partial}{\partial x} \\ \frac{\partial}{\partial y} \end{bmatrix} = [J^{-1}] \begin{bmatrix} \frac{\partial}{\partial s} \\ \frac{\partial}{\partial t} \end{bmatrix} \tag{5.5}$$

where

$$[J] = \begin{bmatrix} \frac{\partial}{\partial s} N_1, & \frac{\partial}{\partial s} N_2, & \frac{\partial}{\partial s} N_3, & \frac{\partial}{\partial s} N_4 \\ \frac{\partial}{\partial t} N_1, & \frac{\partial}{\partial t} N_2, & \frac{\partial}{\partial t} N_3, & \frac{\partial}{\partial t} N_4 \end{bmatrix} \begin{bmatrix} x_1 & y_1 \\ x_2 & y_2 \\ x_3 & y_3 \\ x_4 & y_4 \end{bmatrix} \tag{5.6}$$

and

$$N^T = \frac{1}{4}\,[(1-s)(1-t),\ (1+s)(1-t),\ (1+s)(1+t),\ (1-s)(1+t)] \qquad (5.7)$$

The second derivatives (bending components) are expressed in terms of discrete nodal displacements via an irregular finite difference technique. A two dimensional Taylor series expansion in irregular shaped meshes is employed. This technique has been used in the solution of the large deflection response of a flat membrane, Ref. [6] and the large deflection response of a thin shallow spherical shell, Ref. [7].

Consider a point P_o with coordinates x_o, y_o and five neighboring points $P_i (i = 1, 2, \ldots 5)$ with coordinates x_i, y_i. The Taylor expansion of the normal displacement w results in the following five equations:

$$\begin{aligned} w_i = w_o &+ \frac{\partial w}{\partial x}(x_i - x_o) + \frac{\partial w}{\partial y}(y_i - y_o) + \frac{1}{2}\frac{\partial^2 w}{\partial x^w}(x_i - x_o)^2 \\ &+ \frac{1}{2}\frac{\partial^2 w}{\partial y^2}(y_i - y_o)^2 + \frac{\partial^2 w}{\partial x \partial y}(x_i - x_o)(y_i - y_o) \quad i=1,2,\ldots 5 \end{aligned} \qquad (5.8)$$

where all the derivatives of w are taken at P_o, (Fig. 4a).

In matrix form,

$$\{\underset{\sim}{a}\} = [\underset{\sim}{A}]^{-1}\,\{\underset{\sim}{\Delta w}\} \qquad (5.9)$$

where $$\underset{\sim}{\Delta w}^T = (w_1 - w_o,\ w_2 - w_o,\ \ldots\ w_5 - w_o) \qquad (5.10)$$

and $$\underset{\sim}{a}^T = \left(\frac{\partial w}{\partial x}, \frac{\partial w}{\partial y}, \frac{\partial^2 w}{\partial x^2}, \frac{\partial^2 w}{\partial y^2}, \frac{\partial^2 w}{\partial x \partial y}\right) \qquad (5.11)$$

$$[A] = \begin{bmatrix} (x_1 - x_o), & (y_1 - y_o) & \ldots & (x_1 - x_o)(y_1 - y_0) \\ \vdots & \vdots & & \vdots \\ (x_5 - y_o), & (y_5 - y_o). & \ldots & (x_5 - x_o)(y_5 - y_o) \end{bmatrix} \qquad (5.12)$$

The above expressions can be expressed in terms of the nodal values of w as follows:

$$\{\underset{\sim}{a}\} = [AA]\,\{w\} \qquad (5.13)$$

where

$$\{\underset{\sim}{w}\}^T = (w_o,\ w_1,\ w_2\ \ldots\ w_5) \qquad (5.14)$$

and

$$[AA] = \begin{bmatrix} -\sum_j A_{1j}^{-1} \\ -\sum_j A_{2j}^{-1} \\ \vdots \\ -\sum_j A_{5j}^{-1} \end{bmatrix} \left[\underset{\sim}{A}^{-1}\right] \tag{5.15}$$

Noting that the strain components of interest consist of the last three components of the vector $\{\underset{\sim}{a}\}$, only the last three rows of the matrix [AA] are needed. With the pivotal point P_o labeled as P_k, we now have

$$\{e_k\}_i = [AA_k] \{\underset{\sim}{w}\}_i \tag{5.16}$$

where $\underset{\sim}{w}$ consists of w_k and the displacements at five neighboring points. The selection of these neighboring points is not entirely arbitrary; the following argument reveals the condition which must be imposed on the nodal coordinates x_i, y_i .

The expressions (5.13) and (5.16) yield convergent approximations of the derivatives $\partial w/\partial x . . . \partial^2 w/\partial x \partial y$ provided that the matrix [A] is non-singular. With $x_o = y_o = 0$, the singularity of [A] means

$$\det [A] = \begin{vmatrix} x_1, & y_1 & \cdots & x_1 y_1 \\ \vdots & & & \vdots \\ x_5, & y_5 & \cdots & x_5 y_5 \end{vmatrix} = 0 \tag{5.17}$$

Thus the matrix [A] is singular if there are four such real numbers α, β, γ and δ (not all of them equal to zero) such that

$$x_i + \alpha y_i + \beta x_i^2 + \gamma y_i^2 + \delta x_i y_i = 0 \tag{5.18}$$

for i = 1, 2... 5.

In other words, if the five neighboring points are located on any conic section (ellipse, hyperbola, parabola and two straight lines) the matrix [A] becomes singular. These possibilities rarely occur and can be easily recognized and corrected.

Note, for the case of a rectangular element, the derivative expressions become a staggered finite difference scheme. First derivatives are computed between nodes and second derivatives at nodes.

Two options for choosing elements are available in the EPSA code. An option exists to employ a generalized quadrilateral element which uses the formulation discussed previously. A second option exists to employ a rectangular element which uses a staggered finite difference scheme as discussed in Ref. [4]. A sheet may be discretized with any combination of generalized quadrilateral elements and rectangular elements.

VI SHELL CONSTITUTIVE EQUATIONS - ELASTO-PLASTIC SHELL THEORY DEVELOPMENT

The shell constitutive equations relate the stress resultant rate vector to the shell strain rate vector. This is formulated in matrix notation as

$$\{\dot{s}\} = [D]\,\{\dot{e}\} \tag{6.1}$$

where [D] is the elasto-plastic tangent stiffness matrix.

The shell constitutive equations used differ from the classical elasto-plastic theories in that the formulation involves shell stress resultants rather than stresses at points throughout the thickness of the shell. This avoids the necessity of computing and storing stresses through the thickness of the shell and results in considerable savings in computer storage space and processing time.

However, the stress resultants N_{ij} and M_{ij} of shell theory are not sufficient to describe the state of stress. Certain higher-order moments must be combined with the stress resultants to form the dynamic variables of the problem.

The constitutive relations consist of a yield condition, a strain hardening law and a flow rule.

The stress components at the top and bottom surfaces of a shell are expressed in terms of the stress resultants as

$$\sigma_{ij} = \frac{N_{ij}}{h} \pm \frac{6M_{ij}}{h^2} \tag{6.2}$$

with the plus and minus signs applying to the top and bottom fibers of the shell.

The initial yield surface equation is established by substituting the above relations into Mises yield condition

$$\frac{1}{\sigma_o^2}\left(\sigma_{11}^2 + \sigma_{22}^2 - \sigma_{11}\sigma_{22} + 3\sigma_{12}^2\right) = 1 \tag{6.3}$$

This results in

$$F_o = I_N + I_M + 2|I_{NM}| = 1 \tag{6.4}$$

with

$$I_N \equiv \frac{1}{N_o^2}\left(N_{11}^2 + N_{22}^2 - N_{11}N_{22} + 3N_{12}^2\right)$$

$$I_M \equiv \frac{1}{M_o^2}\left(M_{11}^2 + M_{22}^2 - M_{11}M_{22} + 3M_{12}^2\right) \tag{6.5}$$

$$I_{NM} \equiv \frac{1}{N_oM_o}\left(N_{11}M_{11} + N_{22}M_{22} - \frac{1}{2}N_{11}N_{22} - \frac{1}{2}N_{22}M_{11} + 3N_{12}M_{12}\right)$$

where $N_o = \sigma_o h$ and $M_o = \frac{\sigma_o h^2}{6}$

As the loading of the shell is increased beyond yield, more of the wall cross-section plasticizes until eventually limiting values of the stress resultants are reached. This process is illustrated in Fig. 5 for a simple beam element. In terms of the stress resultants, therefore, the shell exhibits a hardening like behavior even though the material is modeled as elastic-ideally plastic.*)

*) An elasto-plastic material model (hardening in the stress-strain curve) will produce a hardening-like behavior in both the force and moment resultants Fig. 6 . An elasto-plastic material model is currently being incorporated into "EPSA".

A limit surface is constructed by assuming a linear combination of I_N, I_M and I_{NM}. Coefficients for these three terms are determined empirically to produce a satisfactory approximation for the limit surface. The expression

$$F_L = I_N + \frac{4}{9} I_M + \frac{2}{3\sqrt{3}} I_{NM} \tag{6.6}$$

which represents the limit condition exactly for the three special loading cases of (1) membrane forces only, (2) bending moments only and (3) $N_{11} = N_{22}$, $M_{11} = M_{22}$, $N_{12} = M_{12} = 0$ was chosen.

The variable yield condition which describes the "subsequent" yield surfaces as the loading path moves from the initial yield surface toward the limit surface is generated in the following manner. A variable yield surface of the form

$$F \equiv I_N + I_M^* + \alpha I_{NM} = 1 \tag{6.7}$$

is assumed where

$$I_M^* = \frac{1}{M_o^2} [(M_{11}-M_{11}^*)^2 + (M_{22}-M_{22}^*)^2 - (M_{11}-M_{11}^*)(M_{22}-M_{22}^*) + 3(M_{12}-M_{12}^*)^2]^{1/2} \tag{6.8}$$

The residual moments M_{ij}^* represent "hardening parameters" and are defined by the following:

If: $F = 1$ and $\frac{\partial F}{\partial N_{ij}} \dot{N}_{ij} + \frac{\partial F}{\partial M_{ij}} \dot{M}_{ij} > 0$

$$\text{Then:} \quad dM_{ij}^* = 2(1 - F_L) \frac{M_o}{K_o} \frac{F_s^2}{F_M^2} dK_{ij} \tag{6.9}$$

If: $F < 1$ or $\frac{\partial F}{\partial N_{ij}} \dot{N}_{ij} + \frac{\partial F}{\partial M_{ij}} \dot{M}_{ij} \leq 0$

Then $dM_{ij}^* = 0$

The variables F_s and F_M are defined as

$$F_s = [(N_o \frac{\partial F}{\partial N_{11}})^2 + (N_o \frac{\partial F}{\partial N_{22}})^2 + (N_o \frac{\partial F}{\partial N_{12}})^2 + (M_o \frac{\partial F}{\partial M_{11}})^2 + (M_o \frac{\partial F}{\partial M_{22}})^2 + (M_o \frac{\partial F}{\partial M_{12}})^2]^{1/2} \tag{6.10}$$

$$F_M = [(M_o \frac{\partial F}{\partial M_{11}})^2 + (M_o \frac{\partial F}{\partial M_{22}})^2 + (M_o \frac{\partial F}{\partial M_{12}})^2]^{1/2}$$

The formulation of the shell constitutive equations is completed with a statement of the elastic law and the flow rule. Noting that the shell strain vector is composed of an elastic and plastic portion,

$$\underset{\sim}{e} = \underset{\sim}{e}' + \underset{\sim}{e}'' \tag{6.11}$$

where $\underset{\sim}{e}'$ is the elastic strain and $\underset{\sim}{e}''$ is the plastic strain.

The following elastic law is assumed

$$\underset{\sim}{s} = \underset{\sim}{E}(\underset{\sim}{e} - \underset{\sim}{e}'') \tag{6.12}$$

where the elastic matrix $\underset{\sim}{E}$ is the usual shell stiffness matrix.

The plastic strain rates are defined via an associated flow rule

$$\dot{\underset{\sim}{e}}'' = \lambda \frac{\partial F}{\partial s} \tag{6.13}$$

where

$$\lambda = \frac{(\frac{\partial F}{\partial \underset{\sim}{s}}) \underset{\sim}{E} \dot{\underset{\sim}{e}}}{(\frac{\partial F}{\partial \underset{\sim}{s}})^T E(\frac{\partial F}{\partial \underset{\sim}{s}}) - (\frac{\partial F}{\partial \underset{\sim}{s}*}) A(\frac{\partial F}{\partial \underset{\sim}{s}})}$$

and

$$A = 2(1 - F_L) \frac{M_o}{K_o} \frac{F_s^2}{F_M^2}$$

A more detailed discussion of the constitutive theory, including numerical results, is presented in Ref. [3].

VII STRUCTURE-MEDIUM INTERFACE

With the view of establishing in "EPSA" the capability of modeling complicated structures containing internal components, an approxiamtion of the structure-fluid surface interaction which uncouples the acoustic medium from the structure was desirable. Such an uncoupling scheme would allow most of the computer capability to be used in a realistic modeling of the shell and its internal configuration.

The uncoupling procedure used is the Doubly Asymptotic Approximation (DAA) developed by Geers (Ref. [8]) and by Mnev and Pertsev (Ref. [9]). Ranlet et al, Ref. [2]) have employed the DAA method with a natural modal expansion technique to predict accurately the elastic response of ring-stiffened cylinders with internal equipment subjected to shock loading. The DAA therefore has been used to analyze elastic small deflection phenomena.

The inclusion of the DAA into "EPSA" extends the theory into the large deflection, elasto-plastic range.

The nature of many transient structure-media interaction problems is a rapidly applied load followed by a low-frequency response. The DAA produces exact results in the low and high frequency ranges. For short times, DAA reduces to a plane pressure wave approximation and for long times it reduces to a virtual mass approximation.

The DAA imparts upon the structural model surface loading composed of incident and radiated waves. The form of this loading is

$$F_i = A_i p_i + A_i \sum_{k=1}^{M} \frac{Q_k}{\mu_k} \psi_k(s_i) \tag{7.1}$$

where

i) F_i is force imparted on i^{th} node by the fluid.

ii) p_i is incident pressure obtained by an empirical representation of the explosive loading.

iii) μ_k is a coefficient equal to the total wet area of the shell.

iv) $\psi_k(s_i)$ is the k^{th} Surface Expansion Function (S.E.F.) evaluated at the i^{th} node (total of M S.E.F.'s).

v) Q_k is the generalized force in the normal direction for the k^{th} S.E.F.

The generalized fluid forces are expanded along the surface of the shell by means of Surface Expansion Functions (S.E.F.). This technique has been successfully employed by Ranlet et al (Ref. [2]). A direct integration technique for the fluid force is an alternative means of solution which is being considered for the extension of the theory to arbitrary geometries, Ref. [10]. The Surface Expansion Functions represent an orthogonal set of functions over the shell used to circumvent the poorly conditioned behavior of the DAA equations when using expansions in terms of normal mode components (Ref.[11]).

The generalized fluid force is obtained from a system of coupled first-oder differential equations written as;

$$Q_k = \rho_f c \mu_k \{V_{Ik} - \sum_{j=1}^{M} \{\gamma_{kj}\}^{-1} \int_0^t Q_j dt - \frac{1}{\mu_k} \int_A \dot{w}(s_i, t)\, \psi_k(s_i) dA \} \tag{7.2}$$

where

$$V_{Ik} = \frac{1}{\mu_k} \int_A V_I(s_i, t)\, \psi_k(s_i) dA \tag{7.3}$$

V_I is the incident velocity transmitted by the fluid upon the shell. c is the sound velocity in fluid. γ_{kj} is a virtual mass coefficient obtained from the solution of an incompressible steady-state problem in which the normal displacement of each surface expansion function is applied on a cavity having the same shape as the structure.

Equation (7.2) is solved in time with a first-order integration scheme as follows:

$$\{Q\}^{t_i} = (\rho_f c \mu_k) \; [\{V\}^{t_i} - [\gamma] \; \{Y\}^{t_i} - [\Psi] \; \{\dot{q}\}^{t_i}] \tag{7.4}$$

with

$$\{Y\}^{t_{i+1}} = \{Y\}^{t_i} + \Delta t \; \{Q\}^{t_i} \tag{7.5}$$

where the following arrays are defined as:

i) Generalized Fluid Force Vector

$$\{Q\} = \{Y\}$$

ii) Incident Velocity Vector

$$\{V\} = \frac{1}{\mu_k} \sum_{i=1}^{N} V_i(s_i)\psi_k(s_i)A_i$$

iii) Surface Expansion Function Array

$$[\Psi] = \frac{A_i \psi_k(s_i)}{\mu_k}$$

The generalized fluid force is applied to the structural nodes as a dynamic loading.

VIII COMPUTATIONAL FEATURES

The need to realistically model complex structures containing internal equipment has stipulated that efficiency in computation capability be a premium. Therefore, the development of EPSA has proceeded with the goal of being an efficient analytical tool. To this end, basic premises and theories have been introduced. These include:

1) An explicit integration in time scheme, eliminating the need for assembly and inversion of large matrices at each time step.

2) A "Simple" quadrilateral element is used. This introduces a large number of mass points in the structure to achieve an accurate solution for shock and wave propagation problems.

3) The elimination of rotational degrees of freedom which cause stringent stability limitations due to the high frequencies produced by the rotational mass terms.

4) An elasto-plastic theory employing stress resultants, thereby eliminating the need for through the thickness integration.

With the incorporation of these features in EPSA an efficient means of analysis has been achieved.

Coding and computations have been carried out on two versions of EPSA. A small core version which is run on the CDC 6600 machine and a large core version which is used on the CDC 7600 machine.

For the small core version (CDC 6600) computational time is five milliseconds per time step per element. A "typical" problem involving 1000 elements and 500 time steps will consist of a run time of fourty minutes. For each element of the structure, thirty words of data are stored, thereby limited the maximum number of elements per structure to 2500.

For the large core version (CDC 7600) computational time is one millisecond per time step per element. The above "typical" problem therefore consists of a run time of ten minutes. There is effectively no limit on the number of elements per structure.

A re-start provision has been incorporated into EPSA thereby providing the user with an effective means of checking intermediate results.

IX IMPLEMENTATION

During all phases of EPSA development, code verification was enacted to provide a level of confidence and supportability in the code. Subsequent to the inclusion of each new feature in the code's capability, calculations were made to test the code against analytical, experimental and numerical results.

Reference [3] contained checks of the elasto-plastic portion of the code. Included were comparisons of moment-curvature relations using the EPSA shell theory versus the classical through-the-thickness technique. Reference [4] reported on a circular cylindrical shell subjected to small elasto-plastic deformation. EPSA Analysis compared favorably with similar computations using the DYNAPLAS code, Ref. [16].

Analyses were made by EPSA to verify the large deflection capability for both elastic and elasto-plastic stress states due to dynamic loading. Some such results will be presented as follows:

The large-deflection static response of a fixed-ended beam subjected to a point load was investigated, Ref.[12]. Results were obtained assuming (1) linear behavior and (2) nonlinear behavior (large deflections). Problem specifics and results are shown in Fig. 7.

Excellent agreement was obtained with analytical results. The computer model consisted of 6 elements per half length of beam. Computational time was less than two minutes per calculation.

The instability characteristics of a fixed-ended arch was also investigated, Ref. [13]. The magnitude of a concentrated load was varied to determine what the ultimate load capacity of the structure is. Problem specifics and results are shown in Fig. 8. A comparison with a Finite Element Solution by Marcal, Ref. [14], is also shown. Both results are in agreement. The computer model consisted of 6 elements per half length of arch. Computational time was less than two minutes per calculation.

The most worthy determination of whether "EPSA" can indeed provide accurate and reliable predictions for large deflection, elasto-plastic structural response is determined by comparisons of EPSA predictions against relevant experimental data. To this end, the elasto-plastic large deformation transient and permanent response of a circumferentially stiffened cylindrical panel was investigated.

R.W.H. Wu and E.A. Witmer at M.I.T. performed experiments on a integrally stiffened clamped-edge 6061-T6 aluminum cylinder panel subjected to impulsive loading by the sheet explosive loading technique, Ref. [15]. The geometric properties of the panel are shown in Fig. 9. To insure ideally clamped edge conditions, the panel was machined out of an aluminum block leaving a rectangular collar for fixity. For repeatability and confirmation of results three such panels were tested. A sheet of high explosives centered on the upper surface of the panel was detonated providing the impulsive loading.

Analysis of the structure by "EPSA" consisted of a 225 element mesh configuration for the quarter model. The nominal yield of 46 ksi was increased to 60 ksi to account for strain rate effects. Initial conditicns consist of initial nodal radial velocites. These are obtained by equating the impulse imparted by the detonated sheet to the total impulse experienced by the structure. The time-step size used in the integration phase was 0.5 μsec. Total computer run time was less than 15 minutes.

Figure 10 shows comparisons for the deformed panel shape of EPSA versus experimental values. Comparisons of strain time histories are seen in Fig. 11. Excellent agreement was obtained for both deformations and strains.

X CONCLUSIONS

A practical method for predicting the inelastic response of structures in an acoustic medium under dynamic loadings is presented in this report. The "EPSA" code has been developed to serve as an efficient tool which is particularly aimed at meeting the complexities involved in modeling such structures with major internal components. EPSA development has proceeded with computational efficiency as the major objective.

The excellent agreement whcih has been achieved by EPSA in comparison with experimental, analytical and computational results has verified the correctness of the approaches employed in the code.

With this developing level of confidence in the EPSA code, additional capabilities are currently being incorporated. At various stages of development, these are:

(1) An improved multi-sheet capability to provide for the analysis of cylindrical shells with hemispherical or conical end closures.

(2) Inclusion of ambient pressure acting on the shell.

(3) Implementation of a general initial stress and deformation state in the structural model.

(4) Provision for out-of-plane bending of stiffeners (tripping).

(5) A consideration, in the analysis, of major internal structures, etc.

In conjunction with the expansion of EPSA's capability, additional experimental work is being planned. Various scaled and configured models will be subjected to dynamic loadings leading to large elasto-plastic deformations. The behavior of these increasingly complex models will provide a rational basis for the verification of EPSA as additional capabilities are provided in the code.

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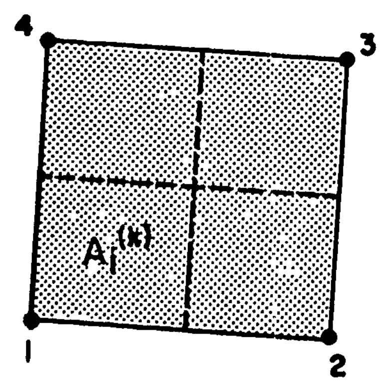
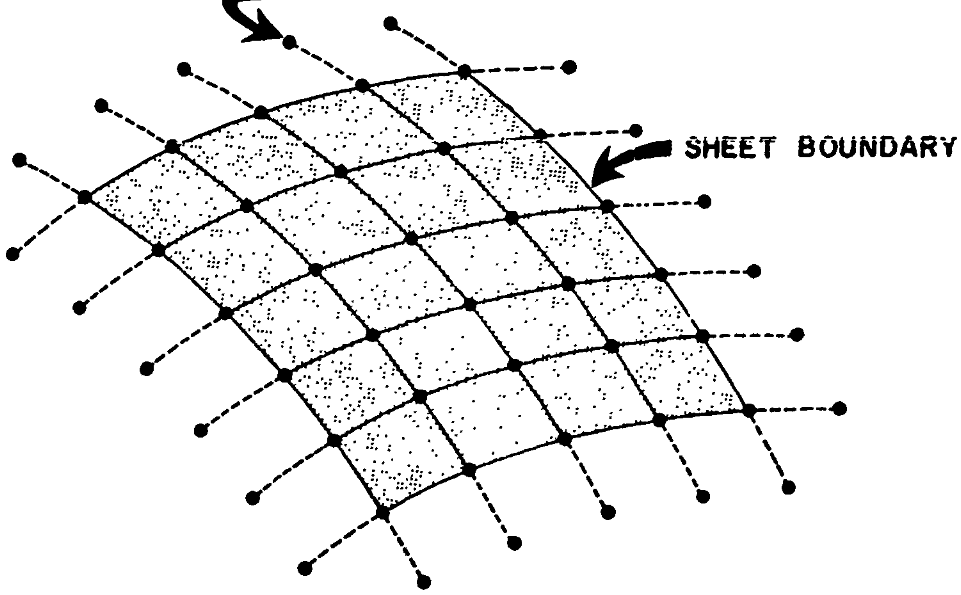


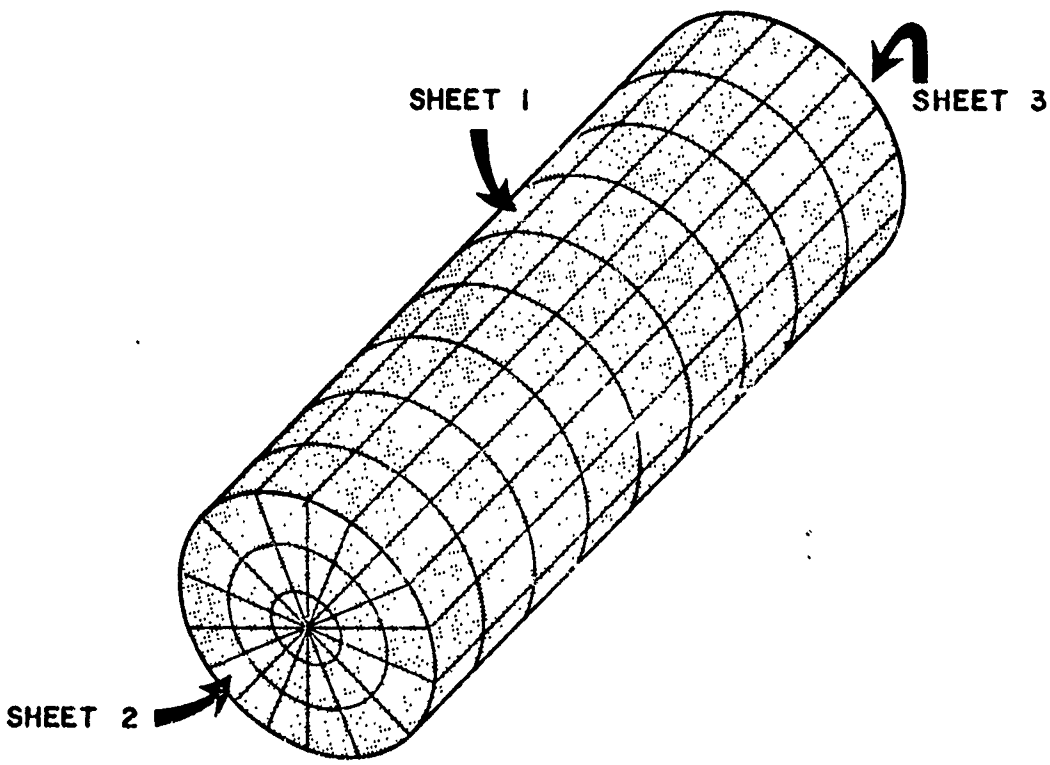
TYPICAL AREA ELEMENT A_i

FIG. 1



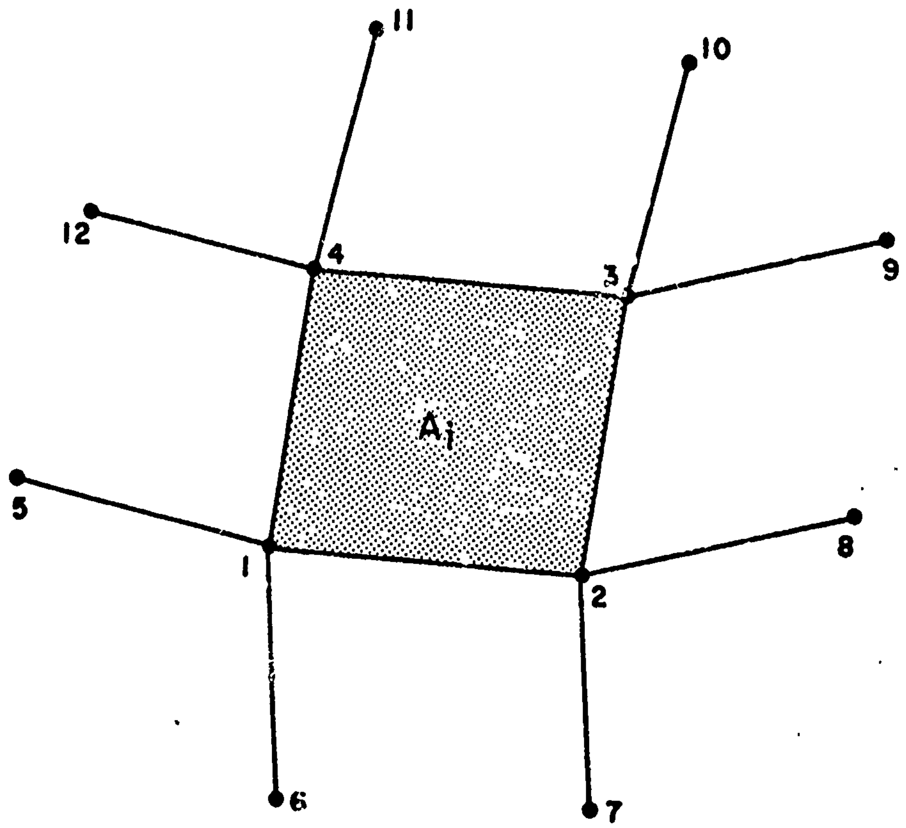
A SHEET

FIG. 2



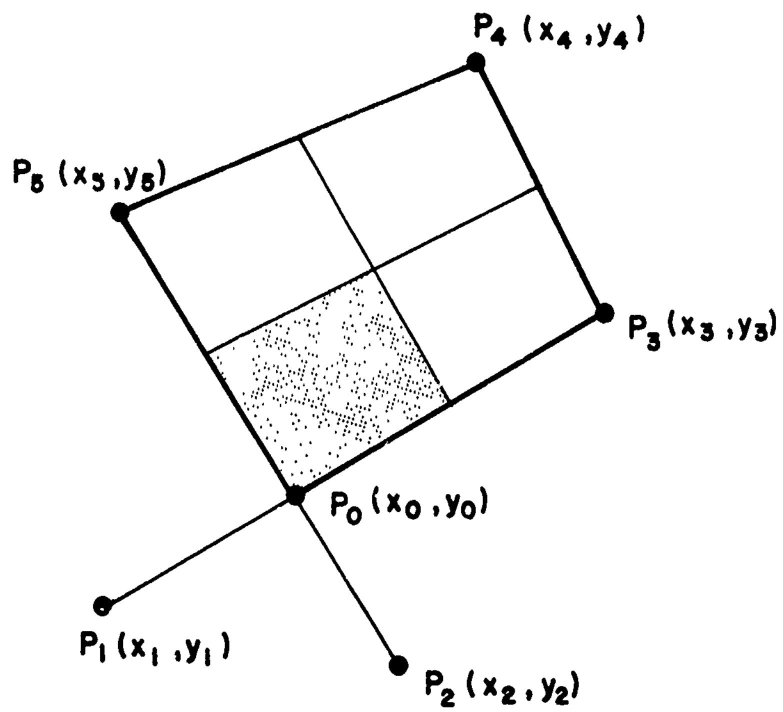
A CYLINDER WITH END CAPS

FIG. 3



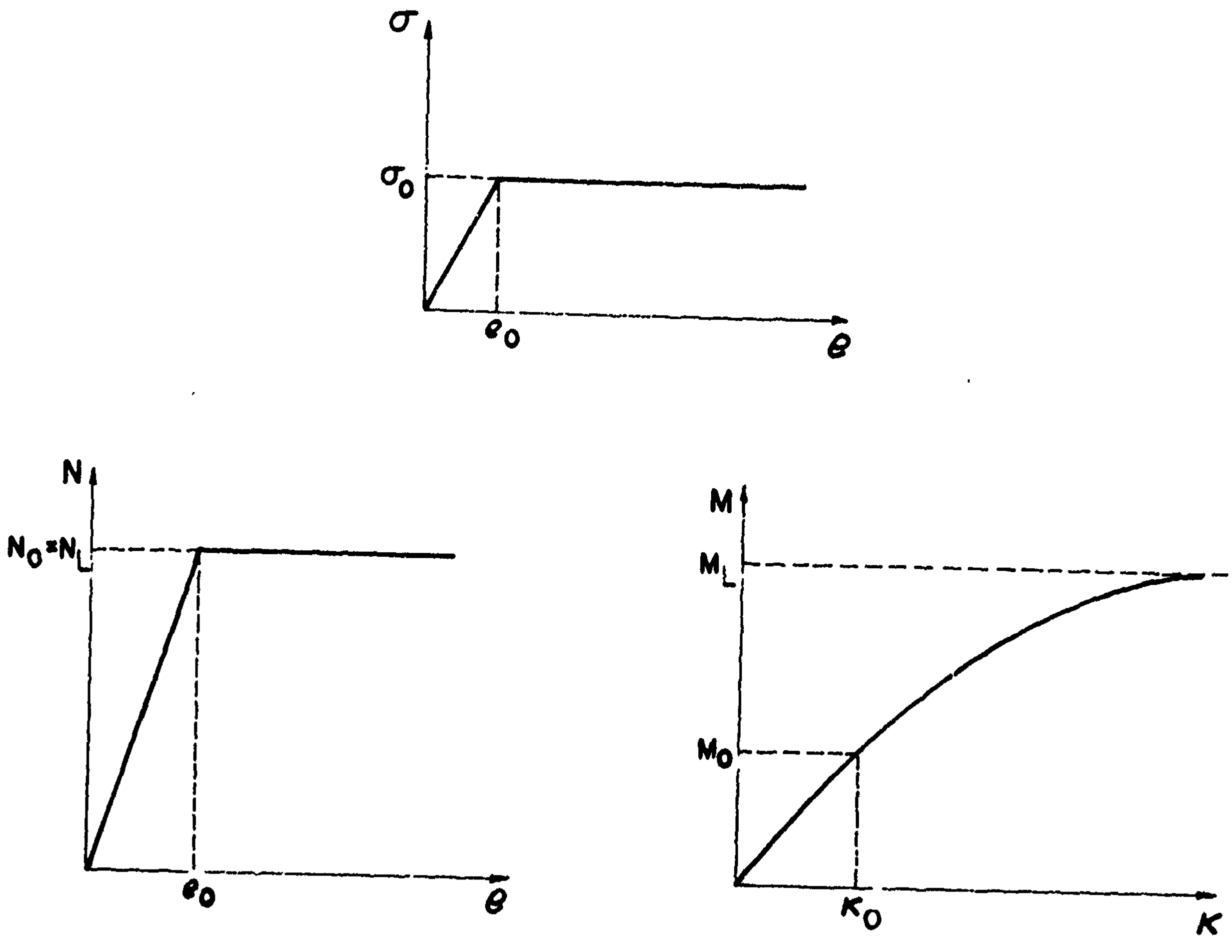
TYPICAL AREA ELEMENT A_i WITH NODAL POINTS USED FOR THE FINITE DIFFERENCE APPROXIMATIONS WITHIN THIS ELEMENT.

FIG. 4



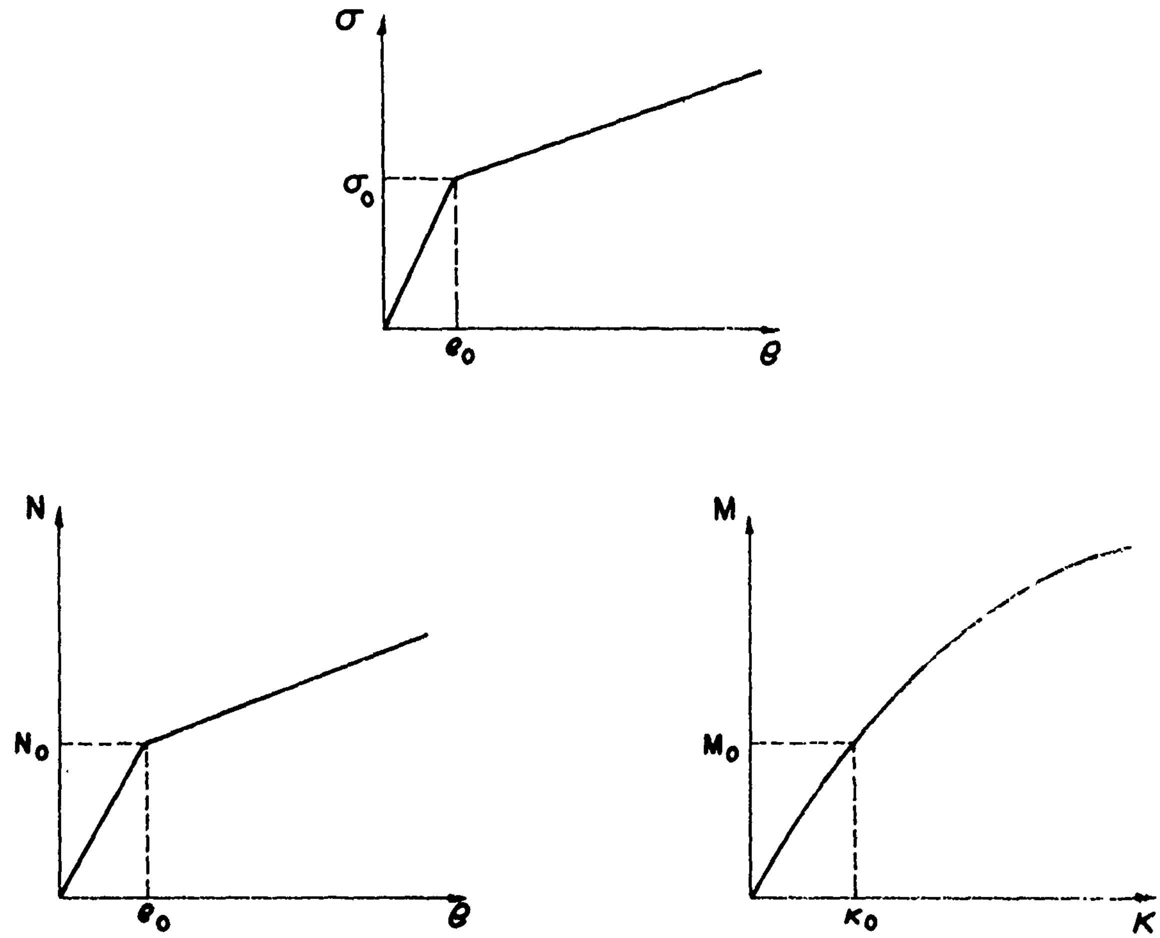
NEIGHBORING POINTS USED FOR TWO-DIMENSIONAL TAYLOR SERIES ABOUT POINT P_0

FIG. 4a



NORMAL FORCE VS. STRAIN AND MOMENT VS. CURVATURE
IN UNIAXIAL STRESS (BEAM BEHAVIOR)
ELASTIC - IDEALLY PLASTIC MATERIAL

FIG. 5



NORMAL FORCE VS. STRAIN AND MOMENT VS. CURVATURE
IN UNIAXIAL STRESS (BEAM BEHAVIOR)
ELASTO - PLASTIC MATERIAL

FIG. 6

STATIC RESPONSE OF FIXED-ENDED BEAM

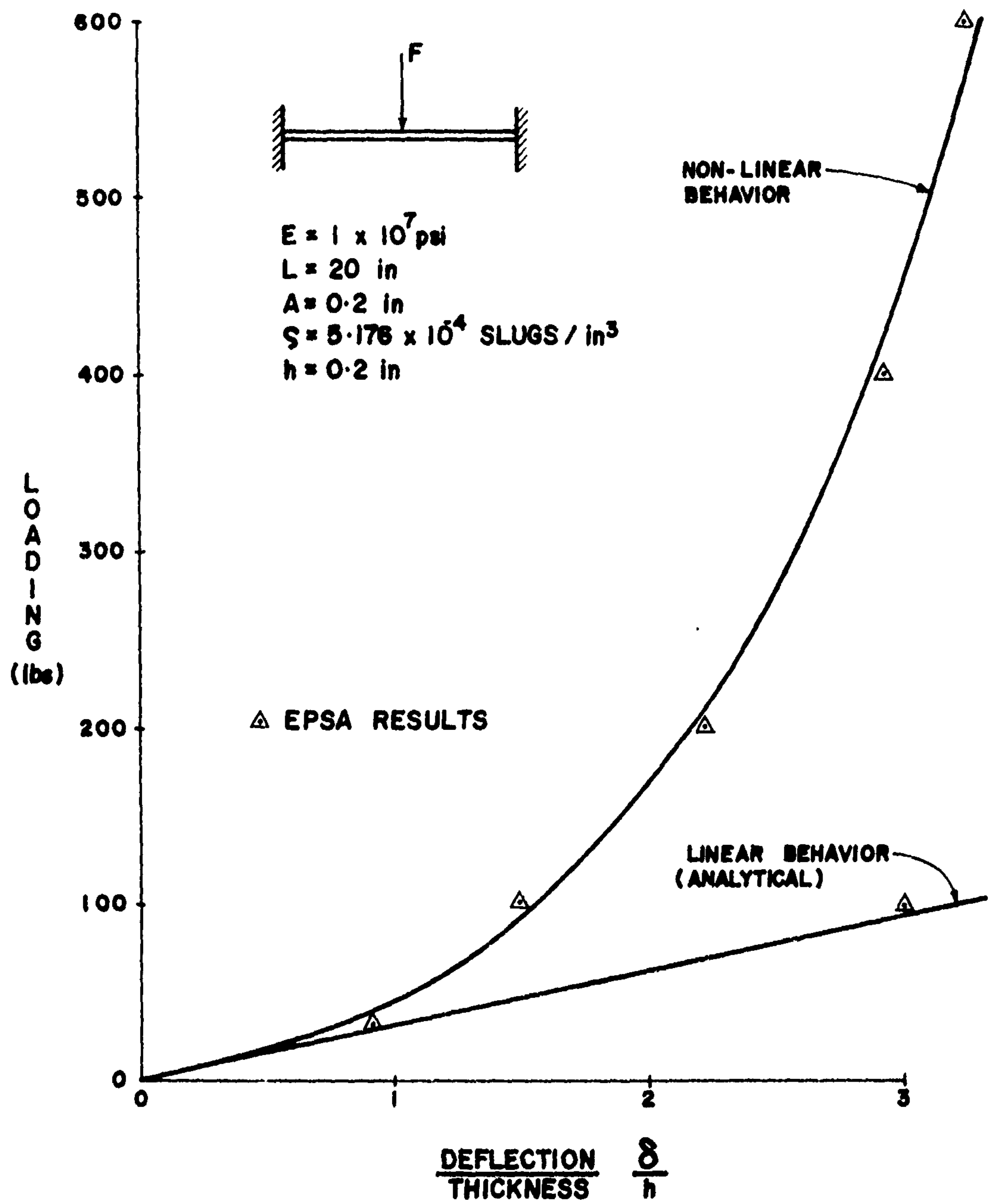


FIG. 7

LARGE DEFLECTION OF ARCH UNDER CENTRAL LOAD

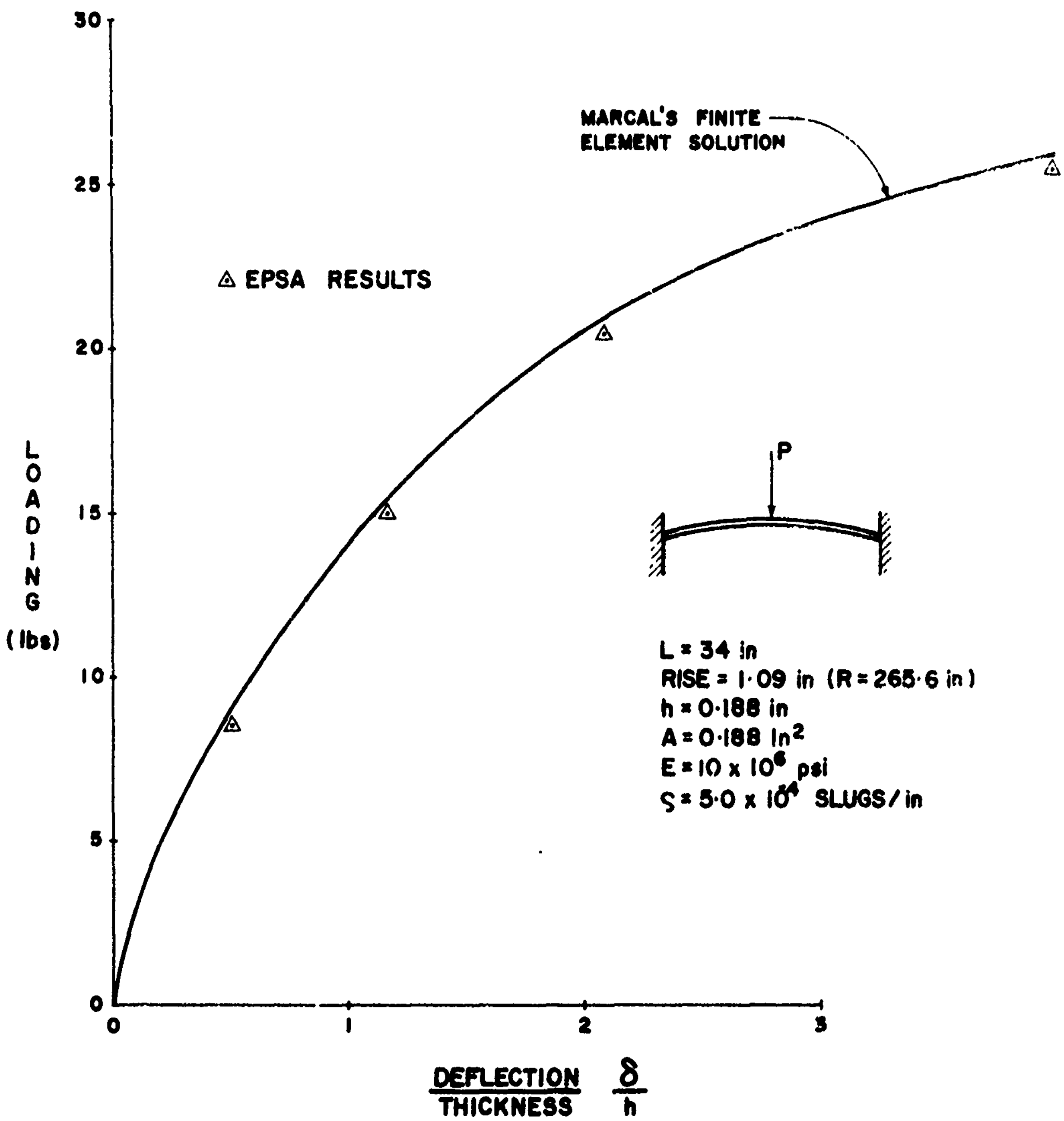
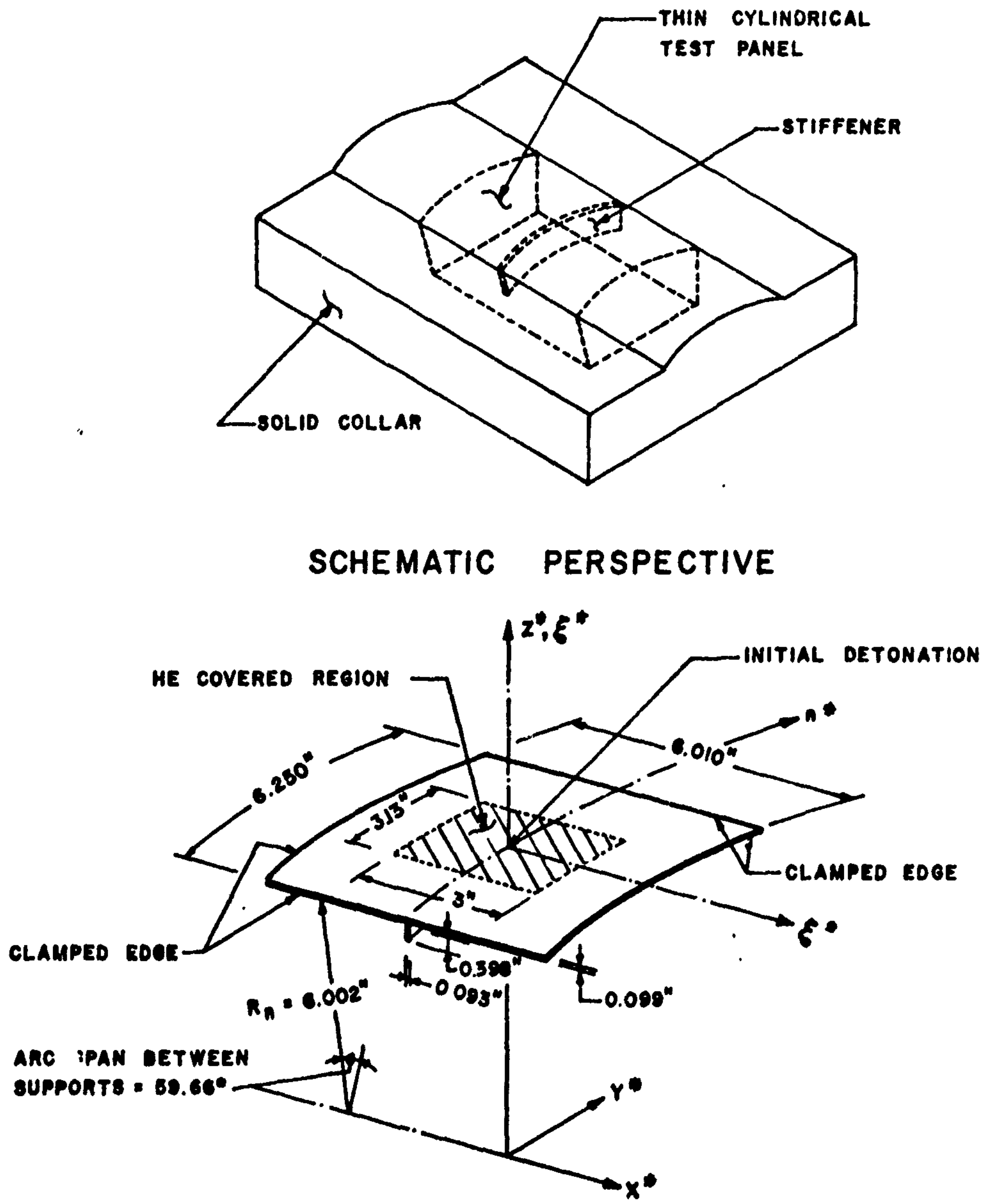


FIG. 8

NON-LINEAR TRANSIENT RESPONSE OF STIFFENED CYLINDRICAL PANEL



DIMENSIONS

SCHEMATICS AND DIMENSIONS OF THE IMPULSIVELY-LOADED, INTEGRALLY-STIFFENED CLAMPED CYLINDRICAL PANEL.

LOAD CONSISTS OF INITIAL NODAL RADIAL VELOCITY $\dot{q}_o$ = 6500 IN/SEC. OVER DETONATED REGION.

FIG. 9

FINAL CROWN LINE DEFLECTION

DEFORMATION PROFILE FOR STIFFENED CYLINDRICAL PANEL

"EPSA" ANALYSIS VS. EXPERIMENT

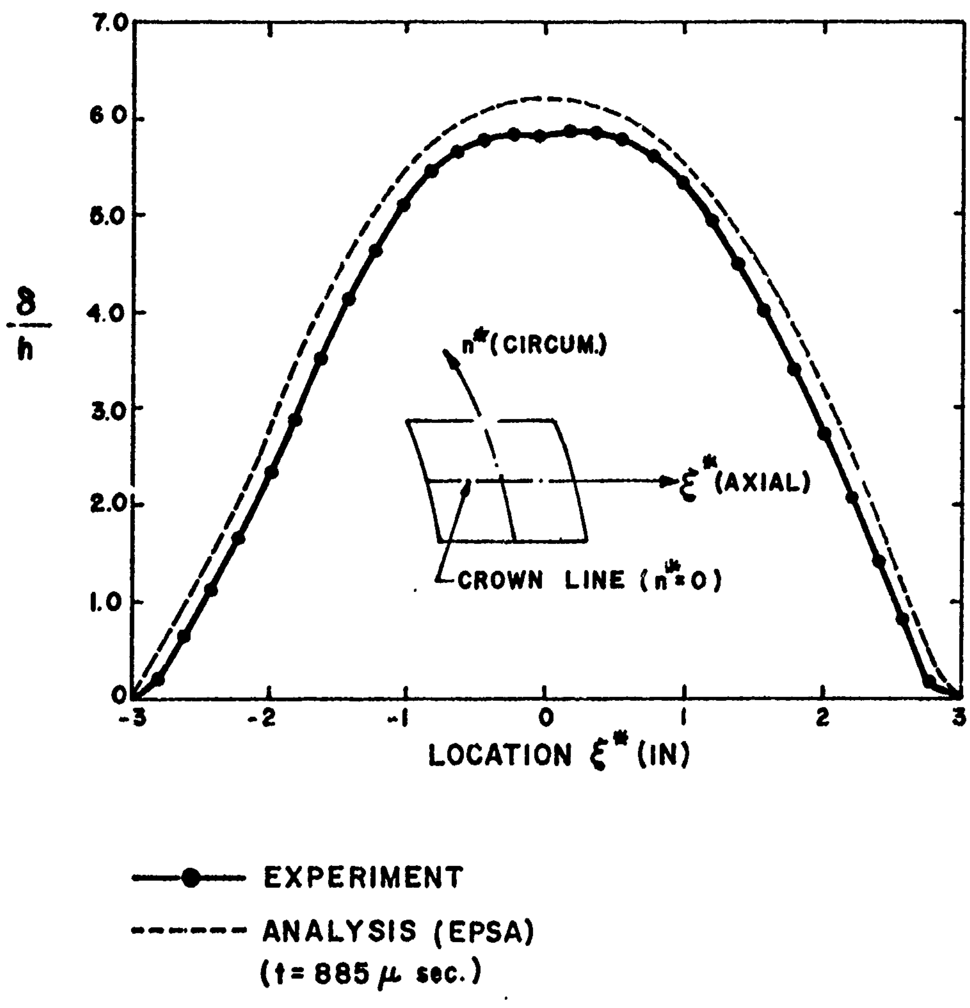
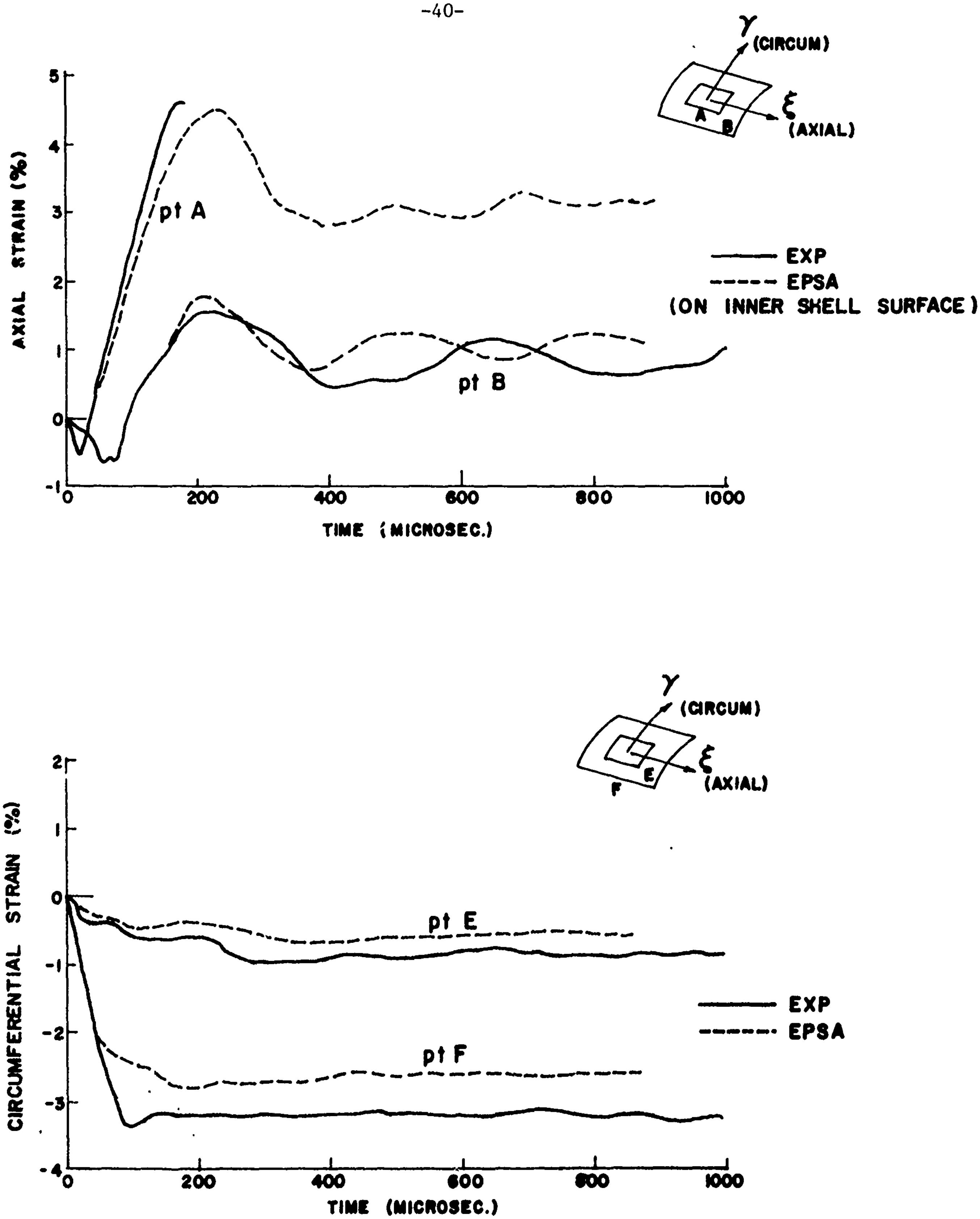


FIG. 10



NONLINEAR TRANSIENT RESPONSE OF CYLINDRICAL PANELS
STRAIN TIME HISTORIES

FIG. 11

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The "EPSA" (Elastic-Plastic Shell Analysis) code has been developed for the analysis of shells in an acoustic medium subjected to dynamic loadings which produce large elasto-plastic deformations in the shell. The analysis includes the modeling of significant internal structures, which produce hard spots on the shell. In addition, the effects of ambient pressure are considered. This report presents the theoretical development for the "EPSA" code and a description of the code itself. A users manual for "EPSA" is planned for the future.

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The structural equations of motion are derived from the principle of virtual work and descretized over the shell in a manner typical of finite element procedures. The integration in time of the equations of motion are done explicitly via a central difference scheme.

The nonlinear Donnell-Vlasov kinematic equations of shell theory are used. Plate strain-displacement relations are established by a two dimensional finite difference scheme.

Two special features have been incorporated into "EPSA" in order to obtain a major gain in the efficiency of the calculations. First, a self consistent plasticity theory for shells has been developed directly in terms of the stress resultants thereby avoiding conventional "through-the-thickness" integrations. Second, a modification of the basic quadrilateral element has been made using finite difference techniques in which the rotational degrees of freedom are removed form the nodal points. As described in the report, both procedures result in a marked increase in computational efficiency, particularly for cases in which large systems are to be analyzed.

The fluid-structure interaction is accounted for by means of the Doubly Asymptotic Approximation (DAA) expressed in terms of orthogonal fluid expansion functions.